NASA TN D-202

NASA TN D-202

7N-34

197734

248.

TECHNICAL NOTE

D-202

EXPERIMENTAL AND ANALYTICAL INVESTIGATION OF

HEAT-TRANSFER CHARACTERISTICS OF A RETURN-

FLOW AIR-COOLED TURBINE ROTOR BLADE

By Francis S. Stepka and Reeves P. Cochran

Lewis Research Center
Cleveland, Ohio

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

WASHINGTON December 1959

(NASA-TN-D-202) EXPERIMENTAL AND ANALYTICAL INVESTIGATION OF HEAT-TRANSFER CHARACTERISTICS OF A RETURN-FLOW AIR-COOLED TURBINE ROTOR BLADE (NASA. Lewis Research Center) 24 p N89-70942

Unclas

00/34 0197734

E-449

CI-1

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

TECHNICAL NOTE D-202

EXPERIMENTAL AND ANALYTICAL INVESTIGATION OF HEAT-TRANSFER CHARACTERISTICS OF A RETURN-FLOW AIR-COOLED TURBINE ROTOR BLADE

By Francis S. Stepka and Reeves P. Cochran

SUMMARY

Comparisons of analytically determined and experimentally obtained blade metal and cooling-air temperatures in a return-flow turbine rotor blade were made to verify the method and assumptions of the analysis. The return-flow air-cooled turbine rotor blade considered herein is a type of blade intended for use in a closed-circuit gaseous-coolant system. The experimental blade temperature data presented were obtained from two test blades that were installed in a modified turbojet engine. The engine was operated at a turbine inlet gas temperature of 1650° F and over a range of blade cooling airflow rates.

Analytically determined average chordwise blade shell and insert temperatures at approximately one-third span from the blade root agreed with experimentally obtained values within 60° and 50° F, respectively.

INTRODUCTION

Turbine rotor blades utilizing closed-circuit coolant systems for gaseous coolants have been considered for application in gas turbine engines (ref. 1). The cooling performance of a typical blade configuration for such a system was determined experimentally and is compared with predicted performance herein. Closed-circuit coolant systems would make possible the use of gaseous coolants such as hydrogen or helium which have excellent heat-transfer characteristics. Turbine blades utilizing these gases can be cooled with less internal coolant surface than is possible with air as the coolant. The closed-circuit system requires a return-flow blade, as discussed in reference 1. In this type blade, the coolant enters at the blade base and is ducted to the tip of the blade and back to a separate region at the base. From a manifold region at the blade base, the coolant is ducted to a heat exchanger for reduction of the coolant temperature and then is returned to the blade. The analysis of reference 1 indicated that the return-flow blade design would cool turbine rotor blades adequately over a wide range of operating conditions using hydrogen, helium, or air as the coolant.

To verify the analysis, an experimental investigation was undertaken to obtain the heat-transfer characteristics of a similar design of the return-flow blade using air as a coolant. The purpose of this report is to present the results of the experimental investigations and to compare these results with those predicted by analytical methods evolved in reference 1. The experimental investigation of the cooling characteristics of the blade was conducted in a turbojet engine that was modified to provide cooling air to two test blades. Cooled blade metal temperature data were obtained at a turbine inlet gas total temperature of about 1650° F and over a range of coolant flows from 0.0084 to 0.038 pound per second per blade (corresponding to coolant-to-gas flow ratios from 0.006 to 0.0274). The cooling-air inlet temperature ranged from 230° to 445° F.

SYMBOLS

A	flow area
$c_{p,a}$	specific heat at constant pressure for cooling air
$D_{h,g}$	hydraulic diameter of gas side of blade, l_g/π
$D_{h,I}$ or $D_{h,II}$	hydraulic diameter of coolant passage, 4A/(wetted perimeter)
F	constant, function of blade transition ratio and Euler number
g	acceleration due to gravity
h	heat-transfer coefficient
J	mechanical equivalent of heat
k	thermal conductivity
l	perimeter
Nu	Nusselt number
n	number of blade spanwise increments
Pr	Prandtl number

E-449

CI-1 back

Re	Reynolds number
$\overline{\overline{Re}}_{g,B}$	average Reynolds number of gas based on average blade temperature, $\frac{\overline{\overline{V}}_g \overline{\overline{D}}_{h,g} \overline{\overline{\rho}}_{g,B}}{\overline{\overline{\mu}}_{g,B}}$
r	radius from center of rotation to root of blade airfoil
T	temperature
V	velocity
w	cooling-airflow rate
Δx	incremental blade or coolant passage length
z	exponent, function of blade transition ratio and Euler number
μ	viscosity
ρ	density
ω	angular velocity

Subscripts:

B	blade or blade shell
e	effective
g	combustion gas or combustion gas side
i	inlet
o	outlet
s	blade insert
x	spanwise distance measured from blade root
Δx	incremental spanwise distance
I	cooling air, inlet coolant passage of blade
II	cooling air, outlet coolant passage of blade

Superscripts:

$\bar{}$ average conditions for an incremental length

$\bar{\bar{}}$ average conditions for entire blade

APPARATUS AND INSTRUMENTATION

Return-Flow Blade

Blade design. - A cutaway drawing of the air-cooled return-flow turbine rotor blade showing the coolant flow path and a photograph of one of the blades used in this investigation are shown in figure 1. Cooling of the airfoil portion of the blade was similar in principle to that of a blade treated analytically in reference 1. The airfoil portion consisted of an outer shell, capped at the tip, and an inner shell or insert (fig. 1(a)). The coolant entered the blade base and flowed into the inlet cooling-air plenum chamber or manifold. From this manifold the coolant was distributed to the annular space between the outer shell and the insert. The coolant flowed radially outward to the blade tip. At the blade tip the coolant reversed direction and flowed radially inward within the center portion of the insert. From the insert the coolant entered a chamber at the blade base and then discharged into an outlet passage within the turbine wheel (see fig. 2).

To adapt such a blade to an available turbojet test engine, it was necessary to design a blade base which, in addition to fastening the blade in the turbine rotor, would provide a coolant inlet and a coolant outlet. Separation of inlet and outlet air was achieved in the blade base by extending the insert through the inlet air manifold (fig. 1(a)). In the radial distance between the base ends of the blade shell and the insert, a manifold was fashioned in the base which encompassed the insert and joined the base of the blade shell for its entire periphery. This manifold was connected to a cooling-air supply by a 3/8-inch-diameter hole in the rear of the blade base. A portion of the lower blade mounting serrations in the center of the base was removed to form a chamber for the discharge of the coolant from the center of the insert. Because the investigation reported herein was concerned with the blade only and not with the auxiliary equipment for the complete closed-circuit coolant system, the cooling air from the exit of the blades was ducted through drilled holes in the rear face of the turbine rotor and discarded into the space between the rotor and the tailcone (fig. 2).

An aerodynamic profile for the blade shell similar to that used in reference 2 was chosen for this blade primarily to expedite the design and fabrication. The thickness of the blade shell was tapered from 0.062 inch at the point of attachment with the blade base to 0.012 inch at the

E-449

E-449

blade tip. The cap for the blade shell was of 0.012-inch-thick sheet stock. At the design operating conditions for this blade in the test engine (11,500 rpm maximum engine speed with a 13-in. tip radius and a 9-in. hub radius) the centrifugal stress in the blade shell at the top of the base platform was 32,200 psi.

In the design of the blade, the profile of the insert was adjusted to give a constant clearance gap of 0.024 inch between the insert and the blade shell, both around the blade profile and over the blade span. The thickness of the insert was tapered from 0.058 inch at the point of attachment with the blade base to 0.010 inch at the tip of the inner shell. The flow areas at the root and tip positions of the space between the insert and the outer shell were 0.0922 and 0.1405 square inch, respectively, and the flow areas through the insert at the same positions were 0.0815 and 0.1737 square inch, respectively. To maintain the centrifugal stresses within the insert to a value comparable with that in the outer shell, it was necessary to terminate the insert at a radius of 12.4 inches. At the design operating conditions in the test engine, the maximum centrifugal stress in the insert was 31,200 pounds per square inch.

Forged Timken 16-25-6 high-temperature alloy was selected for the blade base. The material chosen for the blade shell and the insert was high-temperature alloy HS-25 (L-605).

Blade fabrication. - The blade base was made in two halves. Each half was machined to provide contoured cutouts to fit the ends of the insert and the blade shell, the recess which formed the cooling-air inlet manifold or plenum chamber (fig. 1(a)), and half of the cooling-air inlet hole. The two halves were then brazed together with Nicrobraz to form an integral base structure.

The sheet stock for the insert and the blade shells was taper-ground to the correct thicknesses. From these tapered sheets, halves of both the insert and the blade shell were stretch-formed to the desired profile. The corresponding halves were welded together at the leading and trailing edges over the full span length. The completed blade shell and insert assemblies were then fitted to the base, and the parts were joined together by brazing with Nicrobraz. Grinding of the mounting serrations in the blade base was the next step in the fabrication of the blades. The final operation was the welding of the tip cap onto the blade shell. A photograph of the completed blade is shown in figure 1(b).

Test Equipment

The test engine used in this investigation was a modified production model turbojet engine with a centrifugal-flow compressor and a single-stage turbine. A schematic view of the turbine and tailcone of the test engine is shown in figure 2. Two return-flow blades were placed diametrically opposite each other in the turbine rotor. The engine was installed and operated in a sea-level static test stand.

In order to permit use of the existing test equipment with only minor modifications, the investigation was conducted with air as the blade coolant. The cooling air was introduced to the test blades through an air piping system built into the tailcone and on the rear face of the turbine rotor. Supply lines from a metered high-pressure laboratory air source brought air through tubes in a modified tailcone strut to a 1-inch-inside-diameter tube mounted in the inner tailcone on the engine centerline. Concentric with, and surrounding this tube, was the stationary part of a balanced-pressure sliding ring seal similar to the one described in reference 3. The stationary part of this seal unit consisted of two concentric, spring-loaded, carbon-base rings with a balanced-pressure chamber between them. The rotating part was a chrome-plated disk bolted to the turbine rotor. The cooling air passed through this seal unit and entered a manifold at the hub of the rear face of the turbine rotor. From this manifold, 3/8-inch-inside-diameter tubes, which were attached to the rotor and the blade base, ducted the air to the cooling-air inlet holes in the base of each of the test blades. After circulating through the return-flow blades, the cooling air was discharged through passages in the turbine rotor into the space between the rotor and the tailcone. As stated previously, this investigation was concerned only with the turbine blade; therefore no attempts were made to modify the engine for a closed-circuit cooling system.

Instrumentation

Chromel-Alumel thermocouples were installed on the return-flow blades to measure the metal temperatures of the blades. The locations of these thermocouples are shown in figure 3. On one of the blades, thermocouples were located at the leading edge and the trailing edge of the blade shell at a position one-third of the span length from the base platform to the blade tip. At the same spanwise location but in the midchord region of the other test blade, thermocouples were located on the convex and the concave surfaces of both the insert and the blade shell. To facilitate installation of thermocouples on the insert, provision for thermocouples was made prior to the actual assembly of the blade. This was done by grinding a radial slot in the midchord region on the inside surface of each insert half after it was cut and formed. Tubes were then welded into these slots. After the insert halves had been welded together and brazed to the blade base and the blade serrations had been ground, thermocouples encased in a tube were inserted from the blade base into the tubes on the insert. All the thermocouples on the blade shell were installed after completion of the blade fabrication by grinding slots in the outer surface of the shell and cementing the thermocouple leads in place. The junctions of the thermocouples were spot-welded to the blade shell. A more detailed description of this method of installing thermocouples in thin-walled turbine blades is presented in reference 4.

In addition to the temperatures measured on the test blades, the temperatures of the cooling air at the inlet and the outlet of both blades were measured by thermocouples inserted into the air passages at these points. Thermocouples located at the one-third span, leading-edge position on two of the standard solid blades measured the effective gas temperature relative to the blades.

E-449

The leads from all the above thermocouples were connected to terminal posts located on the rotor hub on the rear face of the turbine. From these terminal posts, the leads extended through drilled holes into the cooling-air manifold at the hub of the rotor, then through a drilled passage on the engine centerline to a slipring thermocouple pickup mounted on the front of the engine. Connections from here to a potentiometer permitted the measurement of the temperatures sensed by the rotating thermocouples.

EXPERIMENTAL PROCEDURE

The blade temperature investigation was conducted in the test engine at rated engine speed (11,500 rpm). The average turbine inlet total gas temperature was about 1650° F. The effective gas temperature for heat transfer to the rotor blades at the one-third-span position was about 1485° F. The cooling air to the blades was varied over a range of coolant flows from 0.0084 to 0.038 pound per second per blade (corresponding to coolant-to-gas flow ratios from 0.006 to 0.0274). Blade metal temperatures, cooling-air inlet and outlet temperatures, and engine gas-flow and coolant-flow rates were measured at each test point. At each test point, the pressure in the balance chamber of the air seal was set equal to the pressure in the cooling-air tube to prevent leakage of cooling air in the seal. Some leakage of cooling air did occur around holes in the rotor hub through which the thermocouple leads were strung. A calibration of this leakage was made in the engine under rotating conditions over the same range of cooling-air pressures as covered in the heat-transfer tests, and the experimental cooling flow data were corrected in accordance with this calibration.

Average experimental blade shell and insert temperatures at the one-third-span position from the root of the blade were determined for use in comparison with analytically determined temperature data. The experimental average blade shell temperature was obtained by weighting the local temperature measurements made at the leading and trailing edges and the midchord regions of the test blades. This weighted average was one-sixth of the following summation: leading-edge temperature plus trailing-edge temperature plus twice the sum of the two midchord temperatures. The use of this relation has satisfactorily determined the average chordwise temperatures of convection-air-cooled turbine blades in which the cooling air was effused from the blade tip. For example, use of this relation and the data of three configurations of cooled turbine blades investigated in reference 5 indicated that the average temperatures obtained were within about 5° F of the average temperature obtained by integrating under curves of the temperature distribution against chordwise distance.

E-449

The insert chordwise metal temperature gradient was assumed to be small because of the insulating layer of coolant between the insert and the hot gas stream. The experimental average insert temperatures used in the comparison with analytical data, therefore, were assumed to be arithmetical averages of the two temperatures measured at the midchord region of the insert.

CALCULATION PROCEDURE

Analysis of Blade Heat Transfer

The blade shell, insert, and cooling-air temperatures at various spanwise positions were obtained from an iterative procedure which was the same as that presented in reference 1. The procedure utilized heat balances between gas and incoming cooling air and between the incoming cooling air and the outgoing air. To solve these heat balances, it was necessary to determine the gas-to-blade-shell, blade-shell-to-incoming-cooling-air, incoming-cooling-air-to-insert, and insert-to-outgoing-cooling-air heat-transfer coefficients.

Gas-to-Blade-Shell Heat-Transfer Coefficient

The gas-to-blade-shell heat-transfer coefficient $\bar{\bar{h}}_{g,B}$ used in the calculation is an average coefficient for the blade and was obtained from the correlation equation of reference 6

$$\overline{\overline{Nu}}_{g,B} = F\left(\overline{\overline{Re}}_{g,B}\right)^{z}\left(\overline{\overline{Pr}}_{g,B}\right)^{1/3} \tag{1}$$

The values of F and z of 0.092 and 0.70, respectively, used herein are the average values of 10 rotor blade profiles including both impulse and reaction blades obtained from reference 7. The gas properties used were evaluated at average blade shell temperature $\bar{\bar{T}}_B$. Equation (1) can be expressed as

$$\bar{\bar{h}}_{g,B} = 0.092 \frac{\bar{\bar{k}}_{g,B}}{\bar{\bar{D}}_{h,g}} \left(\overline{\overline{Re}}_{g,B}\right)^{0.7}\left(\overline{\overline{Pr}}_{g,B}\right)^{1/3} \tag{2}$$

The average gas-to-blade heat-transfer coefficient as determined from equation (2) was used at each spanwise position considered in the calculations.

Blade-to-Incoming-Air, Incoming-Air-to-Insert, and Insert-to-Outgoing-Air Heat-Transfer Coefficients

The heat-transfer correlation equations used to obtain the convective coefficients inside the blade were obtained from reference 8, the data of reference 9, and the assumptions and procedure of reference 1. In the laminar and transition flow regime the heat-transfer coefficients depend on the aspect ratio of the coolant passage, which is the ratio of the height to the width of the coolant passage. The average aspect ratio for the incoming-air passage, of the blade investigated herein, was approximately 96, and that of the outgoing-air passage was approximately 12.5. The following is a tabulation of the heat-transfer-coefficient equations which were used in the laminar, transition, and turbulent flow regimes and the coolant flow Reynolds number ranges in which these flow regimes were assumed to occur:

Flow regime	Reynolds number, $\overline{Re}_{I,B}$	Blade-to-incoming-air heat-transfer coefficient, $\overline{h}_{I,B}$
Laminar	≤ 1240	$4\overline{k}_{I,B}/\overline{D}_{h,I}$
Transition	$1240 < \overline{Re}_{I,B} < 7000$	$0.00322\ \overline{Re}_{I,B}\overline{k}_{I,B}/\overline{D}_{h,I}$
Turbulent	≥ 7000	$0.019(\overline{Re}_{I,B})^{0.8}\overline{k}_{I,B}/\overline{D}_{h,I}$

Flow regime	Reynolds number, $\overline{Re}_{I,s}$	Incoming-air-to-insert heat-transfer coefficient, $\overline{h}_{I,s}$
Laminar	≤ 1240	$4\overline{k}_{I,s}/\overline{D}_{h,I}$
Transition	$1240 < \overline{Re}_{I,s} < 7000$	$0.00322\ \overline{Re}_{I,s}\overline{k}_{I,s}/\overline{D}_{h,I}$
Turbulent	≥ 7000	$0.019(\overline{Re}_{I,s})^{0.8}\overline{k}_{I,s}/\overline{D}_{h,I}$

Flow regime	Reynolds number, $\overline{Re}_{II,s}$	Insert-to-outgoing-air heat-transfer coefficient, $\overline{h}_{II,s}$
Laminar	≤ 2170	$7\overline{k}_{II,s}/\overline{D}_{h,II}$
Transition	$2170 < \overline{Re}_{II,s} < 7000$	$0.00322\ \overline{Re}_{II,s}\overline{k}_{II,s}/\overline{D}_{h,II}$
Turbulent	≥ 7000	$0.019(\overline{Re}_{II,s})^{0.8}\overline{k}_{II,s}/\overline{D}_{h,II}$

Assuming the flow in the incoming-air passage approaches flow between parallel planes, the heat input to the incoming-air passage is primarily from one side only. As a consequence, an assumption is made in reference 1, and used herein, that for flow in the laminar flow regime, the coolant Nusselt number for heat input to one side of a coolant passage is one-half the value of the Nusselt number obtained for equal heat input to two sides. Reference 1 also indicates that the end of the laminar flow regime for heat input to one side only is assumed to exist at values of Reynolds number one-half that assumed for equal heat input to two sides of a coolant passage.

The coolant properties along the blade span were evaluated at the average film temperature for each incremental length of the blade. The film temperature for each of the three coolant heat-transfer coefficients was assumed as the arithmetic average of the respective coolant and metal temperatures at each incremental spanwise length of blade considered.

Spanwise Shell, Insert, and Cooling-Air Temperatures

In the analysis the blade was divided into eight equal increments. The calculation of the temperatures at each of these incremental positions was made using unpublished data on engine conditions corresponding to the operating point at which the experimental blade temperature data were obtained. These unpublished data were average velocities of the combustion gas relative to the blade, average combustion-gas static pressures, and effective gas spanwise temperature distributions. The blade cooling-air inlet temperatures correspond to those obtained in the experimental investigation of the blade.

The iterative procedure for calculating the spanwise temperatures was first to assume an average outside shell temperature $\bar{\bar{T}}_B$, which together with the average velocity of the gas relative to the blade, the average hydraulic diameter of the gas side of the blade, and the gas properties evaluated at $\bar{\bar{T}}_B$, permitted calculation of the gas-to-blade heat-transfer coefficient by use of equation (2). By assuming an average chordwise blade shell and insert temperature $\bar{T}_B$ and $\bar{T}_s$, respectively, for each increment and an exit cooling-air temperature $T_{II,o}$, and by specifying the effective gas temperature distribution, the geometry of the blade, the coolant flow rate, and the cooling-air inlet temperature $T_{I,i}$, the coolant temperatures at the other positions can be obtained. This was done by adding the coolant temperature change due to heat transfer and rotation to the coolant temperature at the beginning of each increment by the use of equations (5) and (6) of reference 1, which in the notation of this report are

$$T_{I,(x+\Delta x)} = T_{I,x} + \frac{\bar{h}_{g,B}\bar{l}_g\Delta x(\bar{T}_{g,e} - \bar{T}_B) + \bar{h}_{II,s}\bar{l}_s\Delta x(\bar{T}_{II} - \bar{T}_s)}{wc_{p,a}} + \left(\frac{\omega^2}{2gJc_{p,a}}\right)(\Delta x)\left[2r + \Delta x(2n - 1)\right] \qquad (3)$$

E-449

E-449

CI-2 back

$$T_{II,x} = T_{II,(x+\Delta x)} - \frac{\bar{h}_{II,s}\bar{l}_s\Delta x(\bar{T}_{II} - \bar{T}_s)}{wc_{p,a}} - \left(\frac{\omega^2}{2gJc_{p,a}}\right)(\Delta x)\left[2r + \Delta x(2n - 1)\right] \tag{4}$$

The assumed values of the average chordwise shell and insert temperature at each increment, $\bar{T}_B$ and $\bar{T}_s$, were checked to agree within 0.1 percent with the calculated values of the temperatures. The equations to determine these temperatures, obtained from heat balances across the outer wall and insert, are

$$\bar{T}_B = \frac{\bar{\bar{h}}_g\bar{T}_{g,e} + \bar{h}_{I,B}\bar{T}_I}{\bar{\bar{h}}_g + \bar{h}_{I,B}} \tag{5}$$

$$\bar{T}_s = \frac{\bar{h}_{II,s}\bar{T}_{II} + \bar{h}_{I,s}\bar{T}_I}{\bar{h}_{I,s} + \bar{h}_{II,s}} \tag{6}$$

The preceding procedure was repeated for each of the eight spanwise increments. At the tip position of the blade the cooling-air temperatures T_I and T_{II} were checked to determine whether they agreed within 2° F; if they did not, another cooling-air outlet temperature $T_{II,o}$ was assumed until this agreement was obtained. The assumed average blade outside shell temperature $\bar{\bar{T}}_B$, required for determining the gas-to-blade-shell heat-transfer coefficient, was checked to determine whether it was within 1 percent of the arithmetic average of the eight spanwise values of $\bar{T}_B$. If agreement was not obtained, another value of $\bar{\bar{T}}_B$ was assumed and the entire procedure repeated. Because of the repetitive nature of the calculations, an automatic computing machine was used.

RESULTS AND DISCUSSION

Experimental Temperature Data

A plot of the blade metal and effective gas temperatures at the one-third-span position from the root of the return-flow blade and the cooling-air inlet and outlet temperatures as obtained in a modified turbojet engine over a range of cooling-air-to-combustion-gas flow ratios from 0.006 to 0.0274 are given in figure 4. The change in the temperatures with coolant-to-gas flow ratio was as expected, except for the trailing-edge temperature. At coolant-to-gas flow ratios of approximately 0.014 the temperature of the trailing edge becomes less than that of the mid-chord pressure surface and continues in this manner as the coolant flow is further decreased. One possible explanation of this variation of the trailing-edge temperatures is that, at the lower coolant flows, a proportionately larger amount of the coolant is directed to the trailing edge as compared with the other regions of the blade because the cooling-air inlet duct to the blade is located at the trailing edge of the blade. Examination of the data indicated that poor cooling of the leading-edge region of the blade was obtained. The remoteness of this region of the blade from the cooling-air inlet duct (fig. 1) probably accounts for the poor cooling effectiveness in this region.

Comparison of Experimental and Analytical Temperatures

The analytically determined spanwise temperature variations of the shell, insert, and cooling air are shown for maximum rated engine conditions in figure 5. Also shown in figure 5 are four data points which were obtained from the data taken at maximum rated engine conditions. Maximum rated engine conditions correspond to an engine speed of 11,500 rpm, a compressor pressure ratio of 3.9, and a turbine inlet total gas temperature of about 1650° F. Other conditions at which the analytical and experimental data were obtained were a cooling-air inlet temperature of 295° F and a cooling-airflow rate of 0.0188 pound per second. The spanwise effective gas temperature distribution relative to the turbine rotor blades in the engine at maximum rated conditions (as obtained from unpublished data) is also shown plotted in figure 5. The data points representing the experimental metal temperatures (fig. 5) are average values obtained by the methods discussed in the section EXPERIMENTAL PROCEDURE. The data points representing the experimental cooling-air inlet and outlet temperatures are the arithmetic averages of the respective temperatures in the two test blades.

The results of the analytically determined blade shell temperatures at the arbitrarily selected coolant flow corresponding to the experimental value of 0.0188 pound per second indicated that appreciable reduction of the metal temperature from the effective gas temperature can be expected. At about the one-third-span position from the blade root ($1\frac{1}{2}$ in. from the root), which is the critical region in the blade shell based on stress-rupture, approximately 335° F of cooling of the shell below the effective gas temperature could be expected. The analysis indicated that the rise in the blade shell temperature along the span was about 270° F. The insert temperature rise along the blade span was about 310° F. The analysis also indicated that a large rise in the coolant temperature (about 470° F) could be expected in the incoming leg of the coolant passage, while the air temperature in the outgoing leg of the coolant passage drops approximately 120° F because of the cooling effect of the air in the incoming leg.

Comparison of the analytically and experimentally obtained shell and insert temperatures at the one-third-span position from the blade root indicated that good agreement was obtained. For the conditions of figure 5, the blade shell temperature agreed within 15° F and the insert temperature within 35° F. The cooling-air outlet temperatures agreed within about 30° F. In the analysis the cooling-air inlet temperature is assumed to be one of the known temperatures and was assumed equal to the experimentally obtained value. A comparison of the analytically and and experimentally determined temperatures over the entire range of cooling airflow at which the experimental data were obtained is presented in figure 6. The results of this comparison indicated that the analytically and experimentally determined values of the blade shell temperatures agreed within 60° F, the values of the insert temperatures agreed within 50° F, and the outlet cooling-air temperatures agreed within 110° F.

E-449

In order to investigate possible reasons for the spread between the experimentally and analytically determined temperatures and to determine whether modification of analytical procedure is warranted, the effect of the errors involved in the use of the equations for the calculation of the gas-to-blade and blade-to-coolant heat-transfer coefficients on blade temperature was considered. The equations used herein for determination of the heat-transfer coefficients were empirically obtained from the experimental data of a number of heat-transfer investigations. The gas-to-blade heat-transfer coefficients determined by the use of equation (2) are within about ±20 percent of the heat-transfer coefficients obtained from the experimental data presented in references 7 and 10 for a group of turbine blades. The blade-to-coolant heat-transfer coefficients determined by the equation for turbulent flow presented herein are within ±10 percent of the experimental data of reference 11 (p. 316) for noncircular tubes having a coolant passage length-to-hydraulic-diameter ratio of 50. Because of this possible spread in the heat-transfer coefficients relative to those obtained by the use of the empirical equations, a spread between the experimentally and analytically determined blade temperatures would be expected. For the maximum-cooling-airflow conditions of this investigation of 0.038 pound per second per blade and for effective gas and cooling-air temperatures at the one-third-span location of 1485° and 375° F, respectively, a maximum spread between the experimental and calculated blade metal temperature at this span location of about ±70° F for a calculated blade metal temperature of 960° F might be expected based on the possible spread in heat-transfer coefficients. Since the spread between the calculated and experimental blade temperatures herein compare favorably with the possible error due to the spread of the heat-transfer coefficients, no attempts were made to refine the temperature calculation procedure. The agreement obtained between the calculated and experimental blade temperatures does not imply that compensating errors between the gas-to-blade and blade-to-coolant heat-transfer coefficients do not exist. However, it was beyond the scope of this investigation to determine experimentally the heat-transfer coefficients for this blade.

Considering only the errors due to the spread of the heat-transfer coefficients, the possible error in predicting the blade temperatures of an assumed high-gas-temperature turbojet engine at high Mach numbers would not change appreciably. At assumed effective gas and cooling-air temperatures of 2200° and 1000° F, respectively, the possible error in predicting blade temperature would be ±85° F for a calculated blade temperature of approximately 1630° F.

SUMMARY OF RESULTS

The results of the experimental and analytical investigations of the heat-transfer characteristics of the return-flow air-cooled turbine rotor blade are as follows:

1. The analytically determined average chordwise blade shell temperatures at approximately the one-third-span position from the root agreed with the experimentally obtained values within 60° F. The analytically determined average insert temperatures at the same span position agreed with the experimentally obtained values within about 50° F.

2. Analysis of errors involved in the calculation of blade temperatures indicated that the results obtained are as good as can be expected with the empirical equations that were used for the determination of heat-transfer coefficients.

E-449

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, September 23, 1959

REFERENCES

1. Slone, Henry O., and Donoughe, Patrick L.: Analysis of Coolant Flow and Pressure Requirements for a Return-Flow Turbine Rotor Blade Design Using Hydrogen, Helium, or Air as Coolant. NACA RM E57B18, 1957.

2. Smith, Gordon T., and Hickel, Robert O.: Preliminary Investigation of Hollow-Bladed Turbines Having Closed and Open Blade Tips. NACA RM E55F27a, 1955.

3. Curren, Arthur N., and Cochran, Reeves P.: A Balanced-Pressure Sliding Seal for Transfer of Pressurized Air Between Stationary and Rotating Parts. NACA RM E56I11, 1957.

4. Stepka, Francis S., and Hickel, Robert O.: Methods for Measuring Temperatures of Thin-Walled Gas-Turbine Blades. NACA RM E56G17, 1956.

E-449

5. Bartoo, Edward R., and Clure, John L.: Experimental Investigation of Air-Cooled Turbine Blades in Turbojet Engine. XII - Cooling Effectiveness of a Blade with an Insert and with Fins Made of a Continuous Corrugated Sheet. NACA RM E52F24, 1952.

6. Brown, W. Byron, and Donoughe, Patrick L.: Extension of Boundary-Layer Heat-Transfer Theory to Cooled Turbine Blades. NACA RM E50F02, 1950.

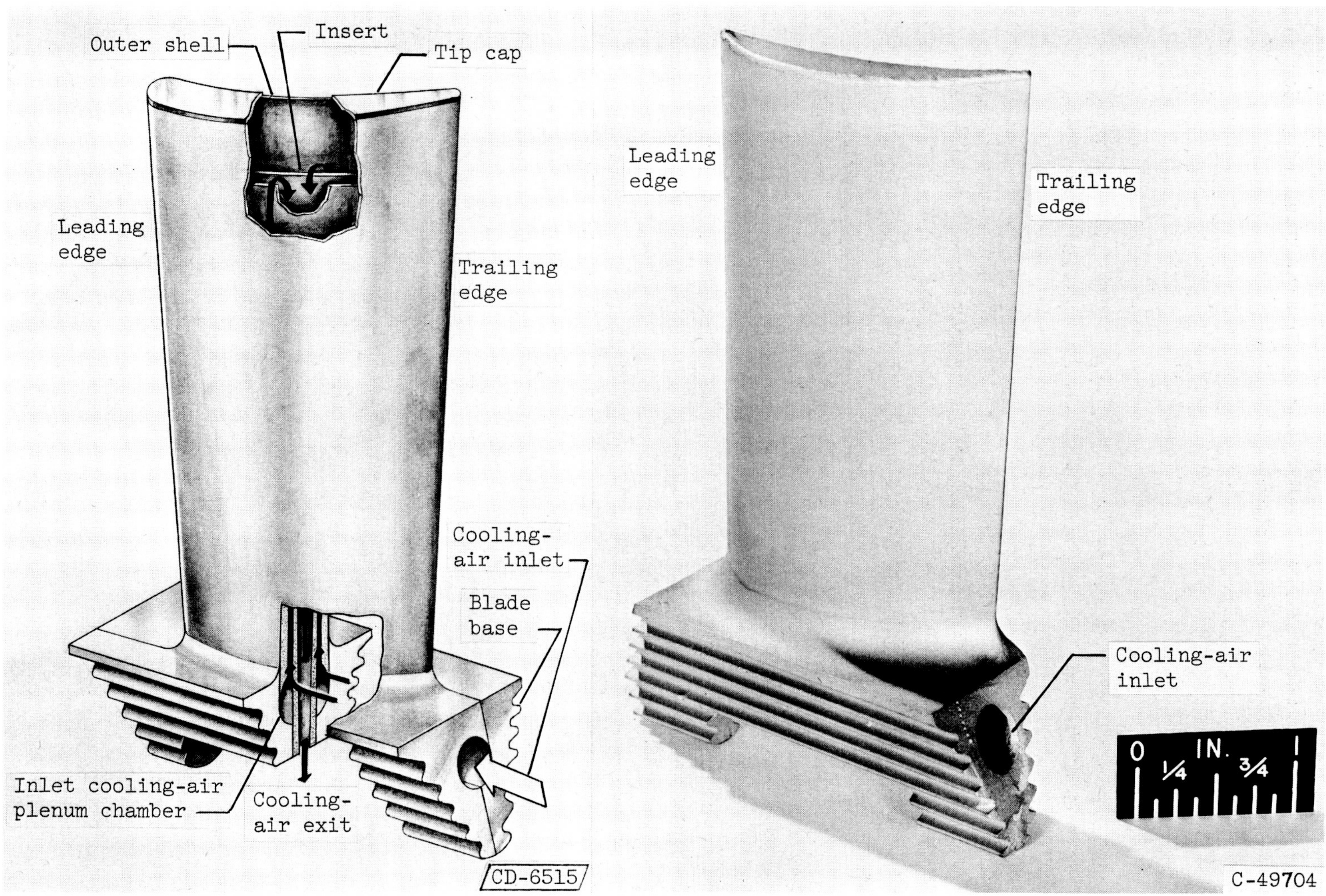
7. Donoughe, Patrick L.: Outside Heat Transfer of Bodies in Flow - A Comparison of Theory and Experiment. M.S. Thesis, Case Inst. Tech., 1951.

8. Ziemer, Robert R., and Slone, Henry O.: Analytical Procedures for Rapid Selection of Coolant Passage Configurations for Air-Cooled Turbine Rotor Blades and for Evaluation of Heat-Transfer, Strength, and Pressure-Loss Characteristics. NACA RM E52G18, 1952.

9. Kays, W. M., and Clark, S. H.: A Summary of Basic Heat Transfer and Flow Friction Design Data for Plain Plate-Fin Heat Exchanger Surfaces. Tech. Rep. No. 17, Dept. Mech. Eng., Stanford Univ., Aug. 15, 1953. (Contract N6-ONR-251 for Office Naval Res.)

10. Hubbartt, James E.: Comparison of Outside-Surface Heat-Transfer Coefficients for Cascades of Turbine Blades. NACA RM E50C28, 1950.

11. Pinkel, Benjamin: A Summary of NACA Research on Heat Transfer and Friction for Air Flowing Through Tube with Large Temperature Difference. Trans. ASME, vol. 76, no. 2, Feb. 1954, pp. 305-318.



(a) Cut-away drawing of blade showing coolant flow path.

(b) Photograph of blade.

Figure 1. - Return-flow turbine rotor blade.

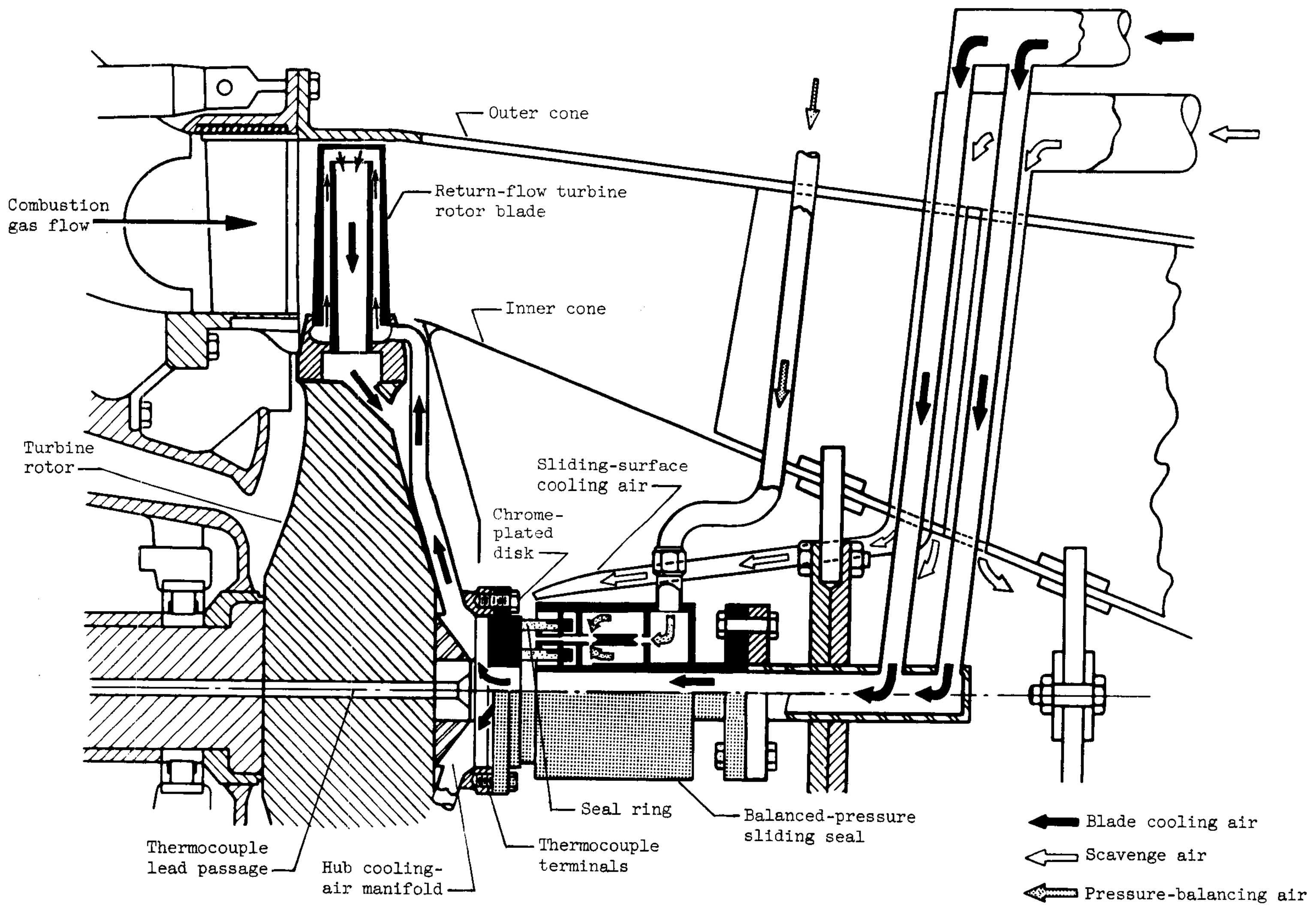


Figure 2. - Schematic drawing of return-flow blade installed in turbine rotor of turbojet engine modified for turbine-cooling research.

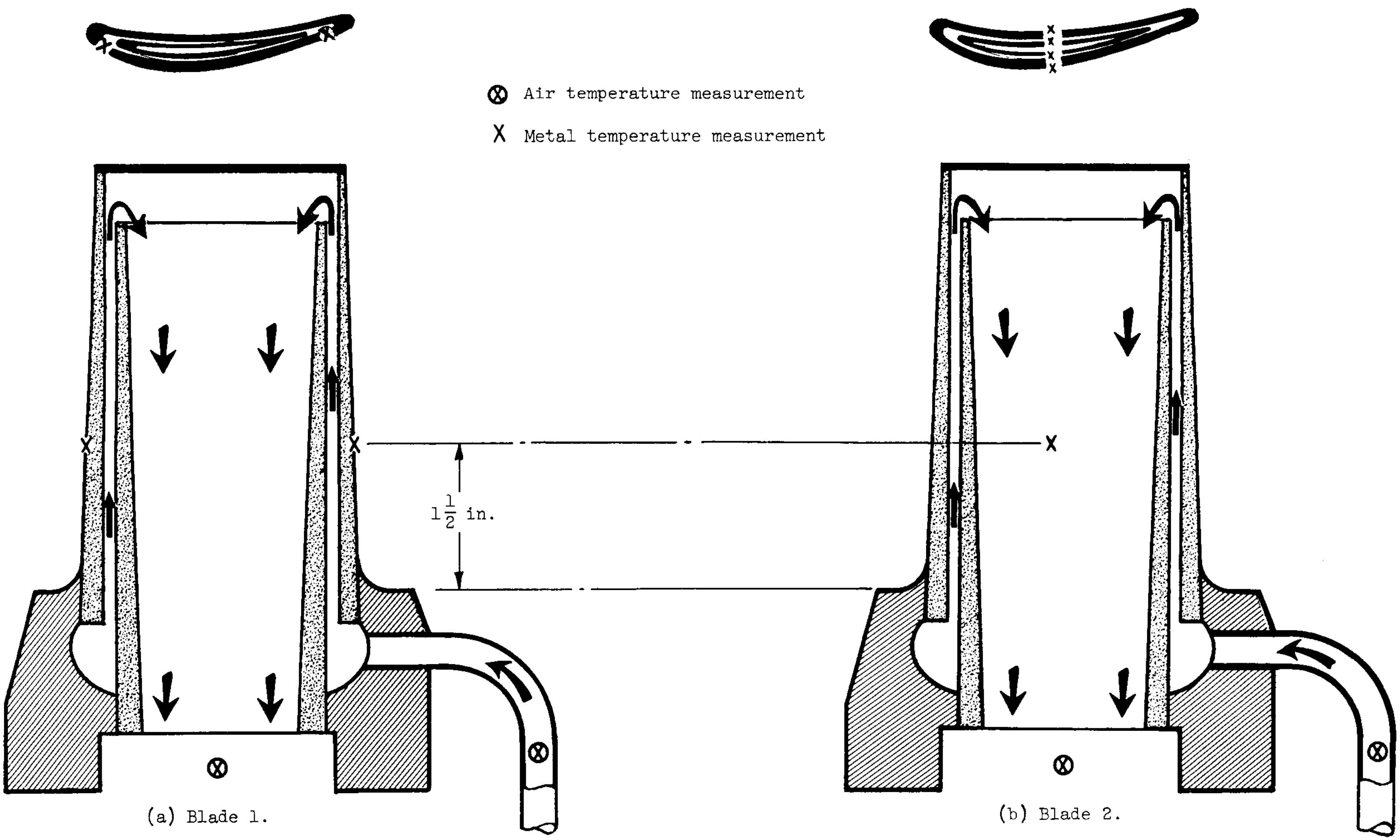


Figure 3. - Location of thermocouples in return-flow test blades.

CD-6516

E-449

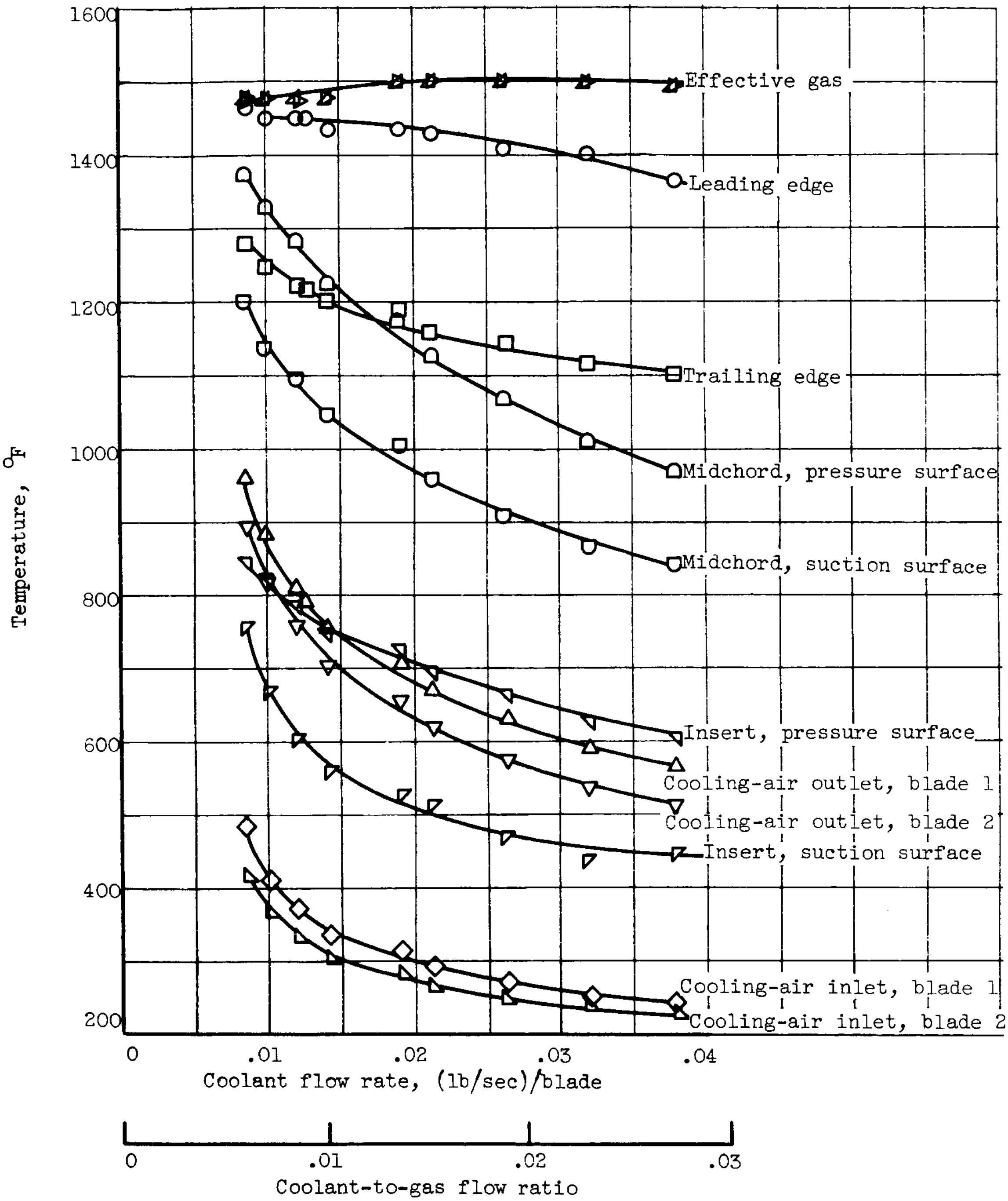


Figure 4. - Blade metal and effective gas temperatures at one-third-span position from root and cooling-air inlet and outlet temperatures over range of coolant-to-gas flow ratios. Turbine inlet temperature, approximately 1650° F; turbine rotor speed, 11,500 rpm.

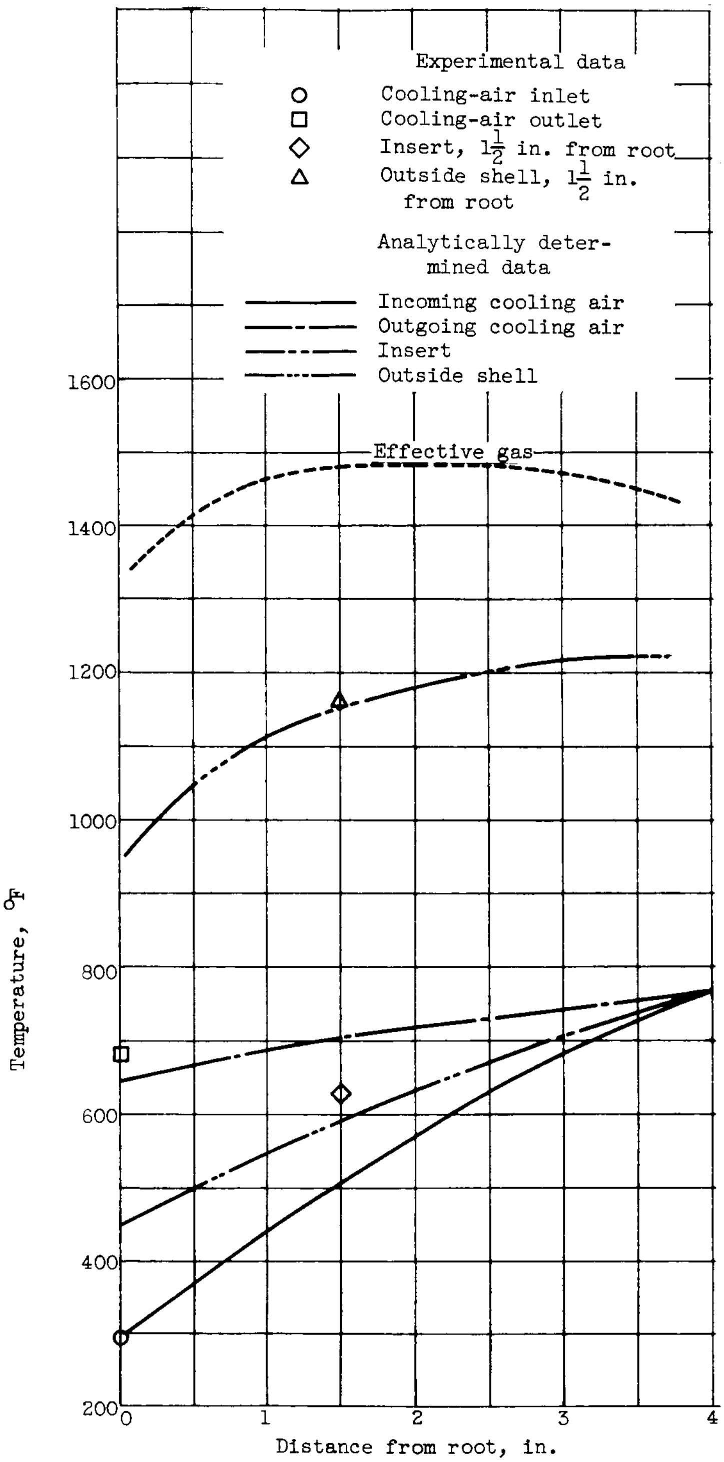


Figure 5. - Calculated spanwise outside shell, insert, and cooling-air temperatures for return-flow blade compared with experimentally obtained data. Total inlet gas temperature, approximately 1650° F; cooling airflow, 0.0188 pound per second; turbine rotor speed, 11,500 rpm.

E-449

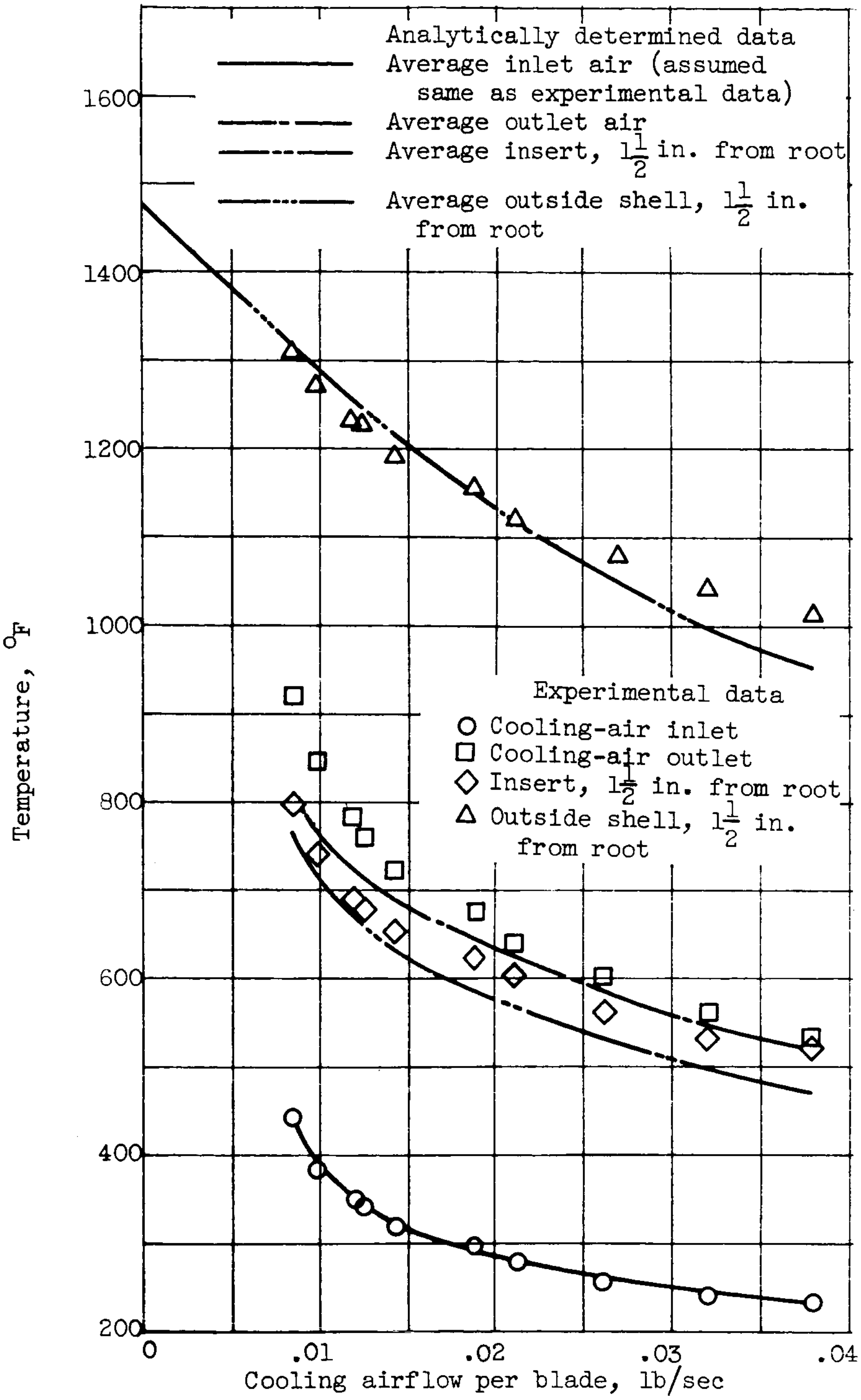


Figure 6. - Comparison of calculated and experimental temperatures of return-flow turbine blade over range of cooling airflows. Maximum rated engine condition: effective gas temperature, 1485° F; turbine rotor speed, 11,500 rpm.